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Preliminary Comparison of Experimental and Analytical Efficiency Results of High-Speed Helical Gear Trains

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PRELIMINARY COMPARISON OF EXPERIMENTAL AND ANALYTICAL EFFICIENCY RESULTS OF HIGH-SPEED HELICAL GEAR TRAINS

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ABSTRACT

An experimental and analytical comparison of the efficiency of high-speed helical gear trains is presented. Analyses of the gearing losses were conducted. Test data from a helical gear train at varying speeds and loads (to 3730 kW (5,000 hp) and 15,000 rpm) was collected. A comparison of the results indicated that the operational conditions of the gearing system affects the loss contributions of the various mechanisms and therefore the overall efficiency of the gear system.

INTRODUCTION

The helical gear train to be studied in this investigation is representative of that required in the prop-rotor gearbox of a tiltrotor aircraft. Tiltrotor aircraft can takeoff like a helicopter and fly like an airplane. The aircraft, shown in figure 1, has tilting nacelles at each end of the wing to facilitate both helicopter and airplane mode operation. During helicopter operation, the nacelles are vertical allowing vertical takeoffs and landings and hover mode operation. During flight, the nacelles can tilt forward, to airplane mode, allowing forward "airplane mode" flight.

The configuration of the aircraft presents interesting challenges to the drive system configuration and design [1]. Each nacelle contains an engine, prop-rotor gearbox and a tilt-axis gearbox. There is a shafting system along the back of the wing that mechanically locks the two rotors together. The aircraft configuration dictated that the engine and mast not have common centerlines (thus making the nacelle shorter in helicopter mode, during landing) requiring the prop-rotor to have a gear train to transfer the power from the centerline of the engine to the centerline of the mast. This is accomplished with a set of high pitch-line velocity helical gears. To maintain weight efficiency, the speed of the gears is maximized to reduce the transmitted torque, thus reducing the gear and bearing sizes.

The high speed design presents a challenge in meeting the aircraft's 30 minute "run-dry" capability for the prop rotor gearbox (where the high-speed helical gear train is used). A complete understanding of the sources, magnitudes, and influences of the heat generated and an accurate method of prediction, will greatly improve the efficiency, accuracy, and quality of the geared system during the design and analysis phases.

PERFORMANCE ISSUES

Many factors can effect the efficient performance of a drive system. For the gears there are several sources of losses. These include the sliding and rolling losses of the gearing action [2–5], for example. The type of lubrication system can also seriously alter the performance of a given gearing system. In aerospace gearing systems, a great deal of trial and error testing is conducted to achieve proper lubricant application as well as scavenging.

Each of these loss mechanisms has operational condition ranges where they can dominate the efficient performance of the gearing system. In the case of low speed operation, the gear mesh losses will dominate the overall performance of the gear system. In high speed gearing systems, this type of loss, while contributing to the system performance degradation, can be a much smaller effect when compared to the windage losses. Data on this loss mechanism, windage, is rather limited in the open literature. Only a few studies have dealt with these issues at high speed [6–10].

In this study, the various gearing performance mechanisms will be separated and their importance described with respect to a high-speed helical gear train. The performance of this type of gearing system was measured in a test facility capable of

Figure 1.—Typical commercial tiltrotor aircraft.

Figure 2.—NASA high-speed helical gear train test facility.

operating to 15,000 rpm and 3,730 kW (5,000 hp) of power. Experiments were conducted to vary the shaft speed and torque applied to the system. These effects will also be investigated analytically using techniques to predict the performance of the gearing members.

TEST FACILITY/TEST HARDWARE/DATA ACQUISITION/TEST PROCEDURE

The test facility designed and fabricated for the study of thermal behavior of high-speed helical gear trains is shown in figure 2. The facility is a closed-loop, torque-regenerative testing system. There is a test gearbox and slave gearbox that are basically mirror images of each other. Each gearbox has an
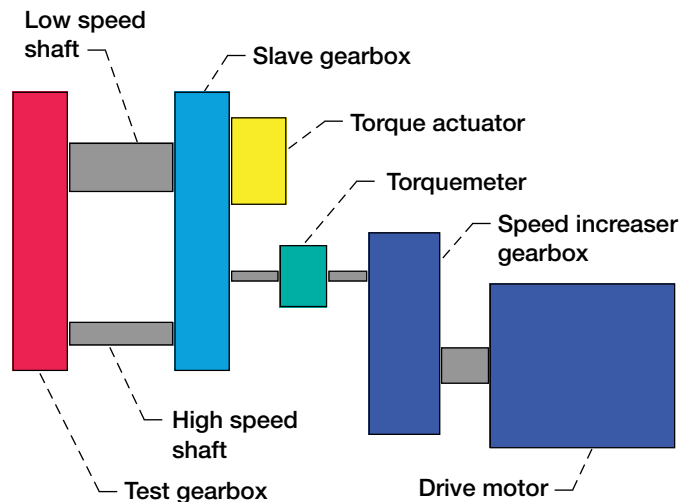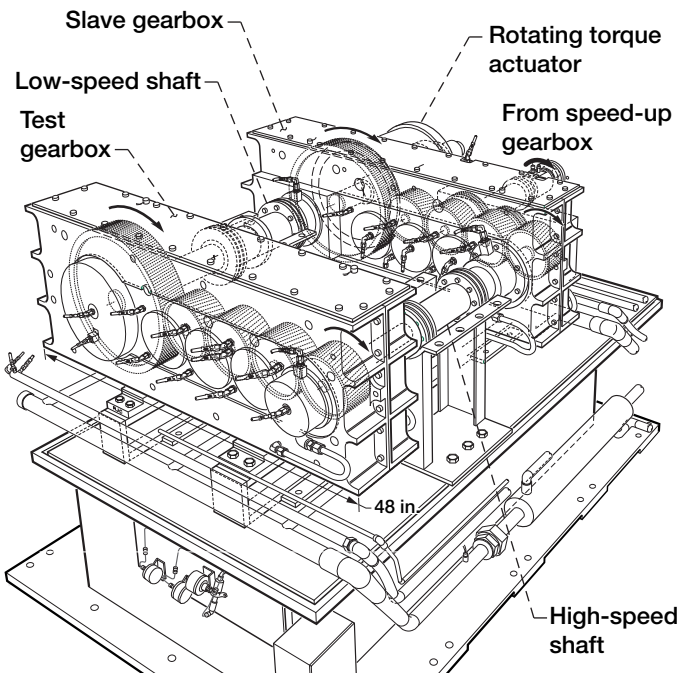

Figure 3.—Layout of NASA high-speed helical gear train test facility.

input gear, three idlers, and one bull gear. The gearboxes are joined together through the input gears and bull gears via shafting.

Within the slave gearbox there is an additional speed increaser section at the first idler. This is the method through which the drive system is rotated and facility power is provided. In this type of facility only the closed-loop losses (friction losses) are necessary to be overcome. Therefore, a drive motor of considerably less power can drive the entire facility. Also within the slave gearbox is a rotating torque actuator that is used to rotate the bull gear in the slave gearbox relative to the shafting from the test gearbox. This ability to rotate the bull gear relative to the shaft permits adjustable loop torque during operation.

The facility is powered by a 373 kW (500 hp) DC drive motor and its output speed is increased using a speed-increasing gearbox. The output of the speed-increasing gearbox then passes through a torque and speed sensor before connecting to the slave gearbox. The entire test stand configuration is shown schematically in figure 3.

Each gearbox has separate supply and scavenge pumps and reservoirs. Lubrication system flow rate is controlled using the supply pressure. Temperature is controlled via immersion heaters in the reservoir and heat exchangers that cool the lubricant returned from the gearboxes. Each lubrication system has a very fine 3-micron filtration. The lubricant used in the tests to be described was a synthetic turbine engine lubricant (DoD-PRF-85734) that is used in gas turbine engines as well as in the drive systems for rotorcraft.

Test Hardware

The test hardware used in the tests to be described are aerospace quality. All components are made of the latest high, hot, hardness gear steels and final ground after heat treatment. The basic gear design information is contained in Table 1. The input and bull gear shafts have bearings to contain the resultant thrust loads, whereas the idler gears only have roller bearings. For the idler gears there is no resultant axial load due to the thrust force balance. There is however an overturning moment, due to the thrust loads, that must be carried by the bearing

Table 1.—Basic Gear Design Data

Number of teeth input and 2nd Idler/ 1st and 3rd Idler/Bull Gear	50/51/139
Module, mm (Diametral Pitch (1/in.))	3.033 (8.375)
Face Width, mm (in.)	67.2 (2.625)
Helix Angle, deg.	12
Gear Material	Pyrowear EX–53

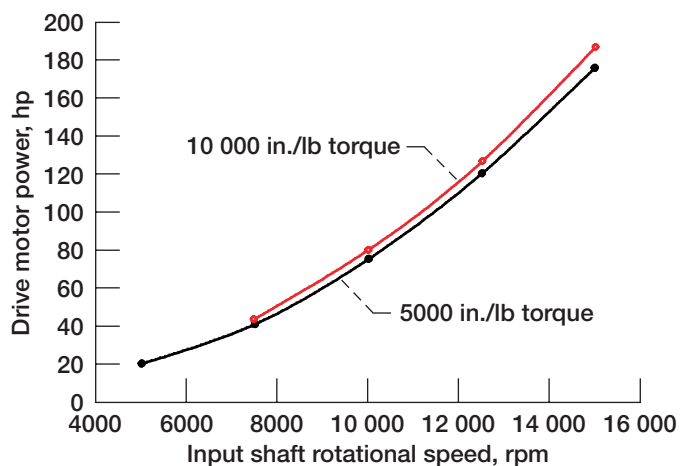Figure 4.—NASA high-speed helical gear train test facility components.

Figure 5.—Drive motor power required to rotate the entire test facility as a function of input shaft speed (bull gear shaft torque shown).
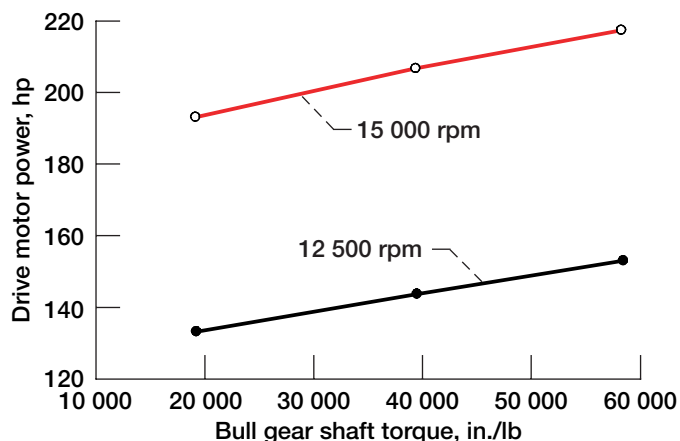

Figure 6.—Drive motor power required at high speed and load.

system. A photograph of the test hardware with the gearbox partially disassembled is shown in figure 4. The bearing inner race is integral to the shafts on the idler gears and at other radially-loaded bearings on the input and bull gear shafts. Shrouds for the gears were used to minimize the windage losses that high-speed gear systems generate.

Data Acquisition

The test facility data system monitors three important facility parameters during operation. Speed, torque (supplied torque and loop torque), and temperature measurements were made during all the testing conducted. The measurement for the supplied torque to the facility is accomplished via a commercially available torquemeter. The test system loop torque is measured on the bull gear connect shaft between the test and slave gearboxes.

The data recording system used in this study has the capability of taking data from all parameters at a rate of one sample per second. The data is displayed to the test operator in real time. Data is stored in a spreadsheet format and each sensor can be viewed at any time during a test or when post-processing the results.

Test Procedure

The test procedure that was followed for collecting the data to be presented was the following: For a given set of conditions, the facility was operated at those conditions for at least 5 minutes or until the temperatures of interest had reached steady state.

EXPERIMENTAL RESULTS

As was shown in the earlier sketches of the test facility, the drive motor power not only supplies the losses for the test gearbox, but it also supplies the losses from the slave gearbox, rotating torque actuator, and an internal speed increaser within the slave gearbox. The amount of power supplied is measured just prior to going into the slave gearbox. The drive motor power supplied at two light torque levels and varying speed is shown in figure 5. As can be seen from this figure, doubling the torque had little effect in comparison to changing the system rotational speed.

Another interesting result is to look at the effect of high torque levels at high operational speed. The results of changing the load from approximately 33 to 100 percent of full torque at two levels of constant speed is presented in figure 6. From this figure, the tests have shown that changing the torque had a linearly increasing effect but was by far much less of an effect when compared to increasing the shaft rotational speed. In both
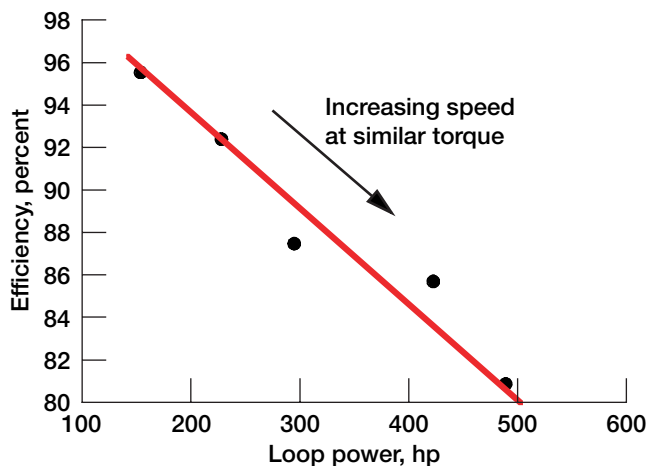
Figure 7.—Effect of shaft speed (input shaft: 5000 to 15 000 rpm) at similar load.
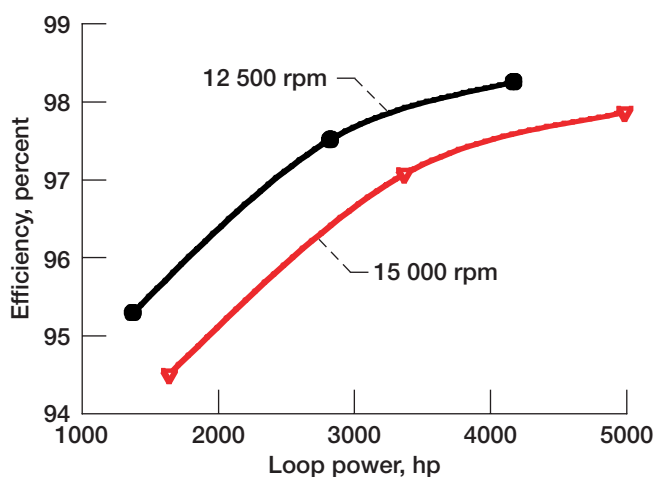

Figure 8.—Effect of high speed and load on helical gear train performance.

Figure 9.—Comparison of supplied power versus calculated power rejected for both the test and slave gearboxes.

figures 5 and 6, the power supplied to the slave and test gearboxes are shown.

Using the data collected during this study, efficiency of the system as a function of conditions can be calculated. The efficiency was calculated using the following measured parameters: loop torque, difference in lubricant temperature from the inlet and outlet locations and jet pressure. The flow rate was established based on the jet pressure provided to 20 jets that lubricate the gears and bearings of the test gearbox. The power losses were the combined effects of the heat rejected to the lubricant and the heat convected to the ambient environment. The test gearbox was modeled as flat plates in a free convection environment. The heat rejected to the oil was typically much greater than that rejected via convection. Only at very low load and speed conditions did the convection portion approach 10 percent of the heat rejection. At the higher loading conditions, typically of interest, the convection part was no more than 1 to 2 percent of the total heat rejection.

The first efficiency comparison that will be made is that of the system operating at relatively light load and varying the
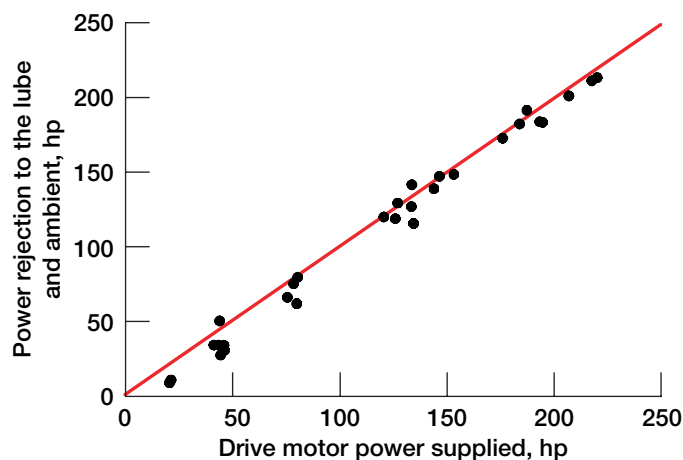
speed. The data used for this comparison was that shown for 9 percent of maximum load. From figure 7, shaft speed had a detrimental effect on the efficiency of the system for nearly constant load.

The next efficiency effect to be looked at is its effect on the helical gear train at higher speeds and varying amounts of substantial load. The effect of load (33, 67, and 100 percent of full torque) and two high-speed conditions (83 and 100 percent of full speed is shown in figure 8. From this figure it can be seen that speed changes from 12,500 to 15,000 rpm has a detrimental effect on the high speed helical gear train performance.

The last comparison that will be made is to look at all the data that had been previously taken and analyzed and determine how well our current method of efficiency performance prediction compares to the power being supplied to the entire geared system. The results of 29 different test points (speed, load, and oil inlet temperature variations) are shown in figure 9.

In this figure, results below the line indicate an under prediction and results above the line indicate an over prediction when compared to the power supplied. Most of the results were under the line. Therefore, the prediction method used in this study may be under estimating the power loss. This difference is believed to be due to experimental accuracy and the assumptions made to analyze the results.

Another way to predict the power loss would be to use the input power required to drive the system and determine how the total power would be split between test and slave gearbox. While not contained in this paper, this was done with the results. If the power loss was assumed to be related to the amount of heat rejected by each gearbox, and the input power split using this assumption, the results given in figure 8 were nearly identical. A comparison of this method to the results shown in figure 7 were different (lower using the input torque splitting discussed above) but followed the same trends.

ANALYTICAL PREDICTIONS

The prediction of gear losses (sliding, rolling, and windage) have been studied by several researchers [3–5]. For the system under study a brief description of the gear member losses will be described.

In any type of gearing there are two main loss mechanisms that occur during tooth meshing, sliding and rolling loss components associated with the relative velocity and loading between the pinion and driven gear. During the meshing action these values vary depending on the location of the line of contact on the profile. Methods have been developed that take into account the gear geometry, speed, and load sharing to predict the losses from the gear mesh.

WINDAGE LOSSES

The windage losses are the least understood and reported type of efficiency degradation mechanism. Very little has been published and the studies that have been conducted were for just a few situations. The analytical tools that are available will be discussed and the methods applied to the geared system under study will be presented.

There are two methods that are similar in form and have been utilized for some time [11]. These two techniques have the same general form, where the power loss is proportional to the diameter to the fifth power and the speed the third power.

The one method only requires three quantities from the design and operation conditions to make an assessment. This equation is the following:

$$P_w = C_1 n^3 D^5 L^{0.7} \tag{1}$$

In this equation C_1 is a constant (1×10^{-17}), n is the speed (rpm), D is the diameter (in.), L is the length of the rotating gear (in.) and P_w is the power loss (hp).

The second method for single helical gears [11], breaks up the calculation into various parts of the gear. The power loss of the gear (assuming a smooth sides) is given by:

$$P_{total} = P_{side} + P_{teeth} \tag{2}$$

where:

$$P_{side} = C_2 n^3 D^5 \tag{3}$$

and

$$P_{teeth} = C_3 n^3 D^4 L \left(\frac{R_f}{\sqrt{\tan \varphi}} \right) \tag{4}$$

In these equations C_2 and C_3 are constants, R_f is related to the "pitch or coarseness" of the teeth and φ is the helix angle.

The other current source of windage work has been that conducted by Dawson [7,8]. In these two papers, experimental tests of windage were conducted on a number of gear diameters, pitches, axial length and to some extent environmental effects. An equation based on the results was developed as the following:

$$P = C_4 C' \rho n^{2.85} D^{4.7} v^{0.15} \lambda \tag{5}$$

In this equation C_4 is a constant (1.12×10^{-8}), C' is a value dependent on the face width to diameter ratio and the number of teeth on the gear, ρ is the density of the gear operational environment (kg/m³), n is the gear rotational speed (rpm), D is

the diameter of the gear in question (m), v is the kinematic viscosity of the operational environment (m²/s), and λ is a constant related to the type of housing/shrouding surrounding the gear (λ = 1 (open); = 0.7 (loose enclosure); = 0.5 (close fitting case)).

As can be seen from the equations above, all have roughly the same form with speed to the third power and diameter to the fifth power. The later study conducted by Dawson [8] has initiated the more difficult portion of the problem, that is, what to do about the ambient environment (oil-air mixture) and how to deal with the gearbox casing (shrouds, windage trays, etc.).

In equations (1) and (2) there is no environmental effect and gear size (diameter) will dominate the results. For the gear system of this study the results are shown in figure 10 for eq. (1) and figure 11 for eq. (2).

Using the work of Dawson, the results from the components in this study are shown in figure 12. The power loss contribution from the gears are switched or the large gear dominates the results in figures 10 and 11 whereas in figure 12 the higher-speed, smaller-diameter gears have a higher power loss.
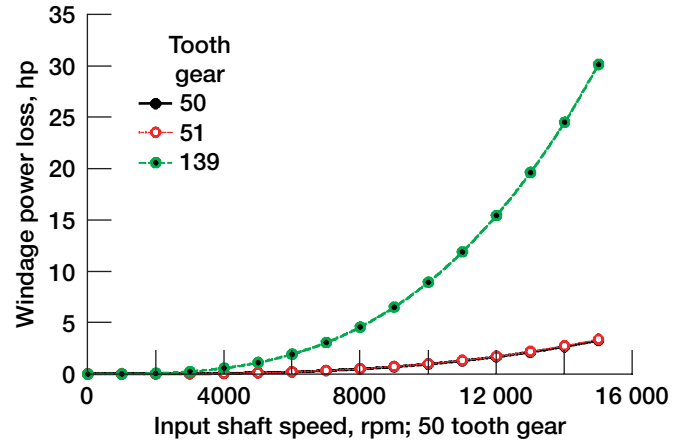

Figure 10.—Windage power loss as a function of speed and gear type (Eq. (1)).
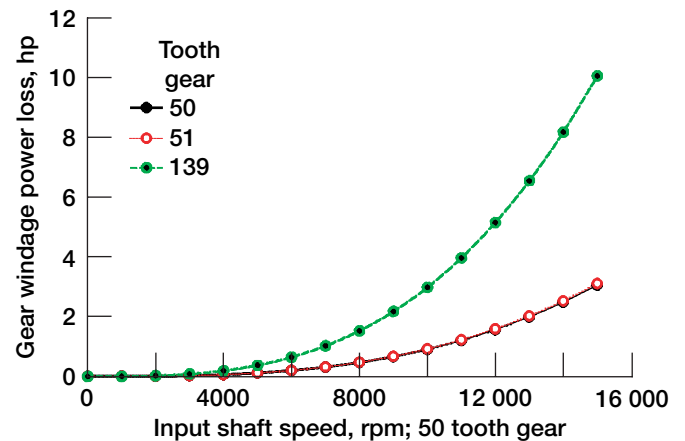

Figure 11.—Windage power loss as a function of speed and gear type (Eq. (2)).
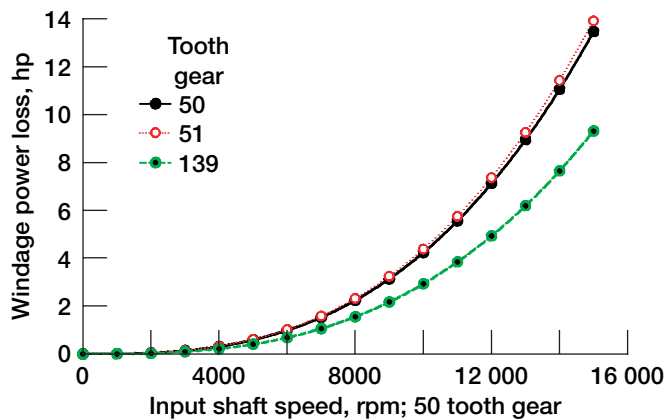
Figure 12.—Windage power loss as a function of speed and gear type from Eq. (5) (1% oil mixture in the ambient air, and an enclosure factor λ = 0.5).
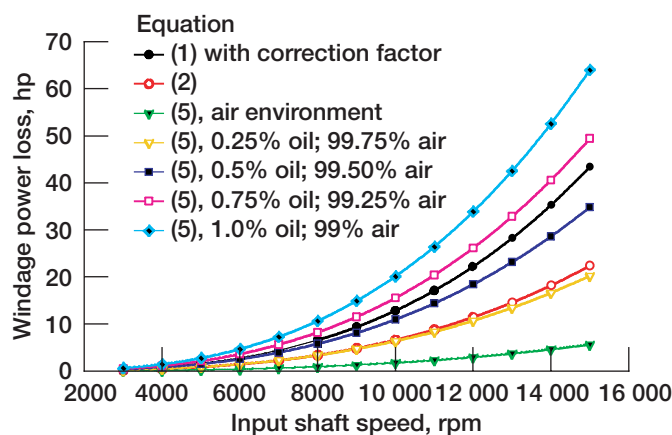

Figure 13.—Windage equations plotted versus speed for the entire gear train.

Taking a look at the entire gear train (input gear, 3 idlers, and a bull gear) the net windage power loss from each analysis is shown in figure 13.

As can be seen from this figure, the environment has a significant effect on the net windage power loss as calculated using eq. (5). Also, from this figure, the effects of windage are of no real concern no matter which equation is used until approximately 8,000 rpm. At this point the pitch line velocity is approximately 53.5 m/s (12,500 ft/min) for the helical drive train of this study.

Therefore, at the present time it is difficult to pick an equation, method, or even the essential information needed to perform some of the calculations. This windage calculation along with losses found in the gear and bearing losses, should allow us to make an estimation of the parameters for a given system.

GEAR MESHING LOSSES

In this paper, the results pertaining only to the gearing components were analyzed for the same conditions as are shown in figures 7 and 8. In the gear train considered in this study, there are a total of four different gear meshes. The layout of the gearing arrangement is shown in figure 14. In a gear train
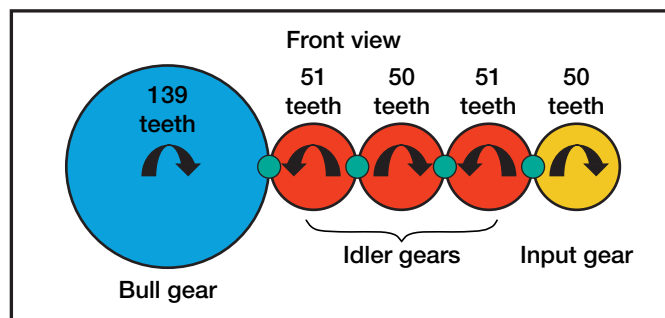

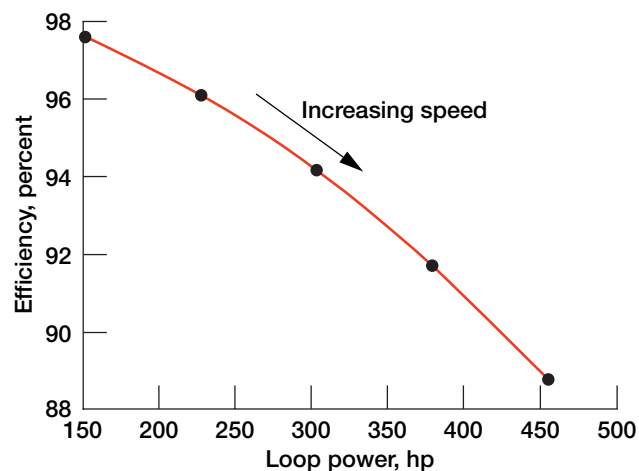Figure 14.—Layout of helical gear train arrangement.

Figure 15.—Analytical prediction of the high-speed helical gear train efficiency.
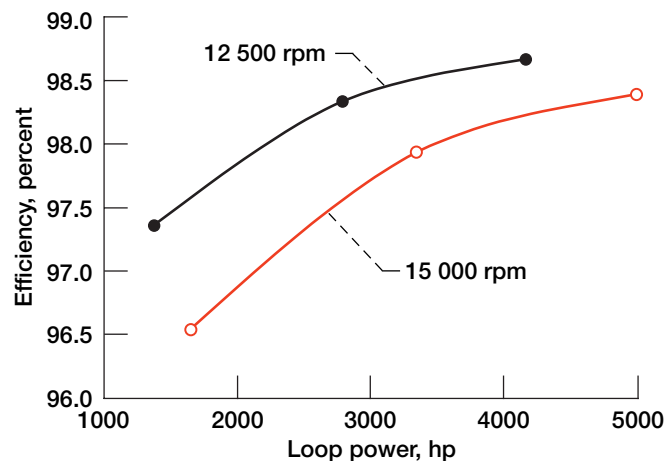
Figure 16.—Analytical prediction of high-speed helical gear train efficiency at high speed and load.

such as this, the idlers each are involved in two meshing actions.

Therefore, the gear meshing losses were calculated for the 50 tooth to 51 tooth mesh twice and the two other meshing arrangements (51 tooth to 50 tooth, and 51 tooth to 139 tooth). The sliding and rolling losses for each of the four meshes and the windage losses from each of the five gears were combined in the analytical results shown in figures 15 and 16.

In figure 15 the efficiency of the gear system is shown at light and constant load. As the speed is increased, the efficiency decays due to the drastic effect of windage at the light load condition.

In figure 16 the effect of varying the level of load at two high speed conditions is shown. Increasing the speed at the constant torque levels lowered the gear train efficiency.

DISCUSSION OF RESULTS

In both the experimental and analytical results presented, the high speed helical gear train efficiency was adversely effected by the rotational speed of the gearing system. At the higher speeds and loads the windage power loss can nearly equal or exceed that due to the gear meshing action. Therefore, substantial improvements in the gear system performance can be made if the windage losses could be reduced. Gear design changes may slightly effect the windage loss and can have an effect on the gear meshing losses, but techniques that would reduce this power loss component would be beneficial whatever the high speed gear design.

For the highest speed and load conditions, the gear mesh and windage losses were predicted to be 80 hp (at 15,000 rpm, 20,935 in.•lb torque). The experimentally measured losses for the same conditions were approximately 98 hp. An improved simulation is possible, but requires assessing the shafting windage and the bearing losses (not included in this report). However, the analysis appears to produce similar trends as that of the actual tested performance.

The extremely difficult portion of the analysis is how to improve on the windage prediction and more important is what steps are necessary to improve the overall efficiency. There are so many parameters such as gear design, housing arrangement, shrouding, and lubrication system effects that a concise analytical representation may not be possible. One way to continue performance improvements is to conduct additional experimental comparisons where one variable at a time is studied. This will lead to trends for improved performance and windage reduction.

CONCLUSIONS

Based on the testing and analysis that was conducted in this study the following conclusions can be drawn:

1. High gearing system rotational speed has a drastic effect on the efficiency of high speed helical gear trains.

2. Both experimental results and analytical predictions demonstrated that windage losses will dominate the performance when light loads and high speed is applied to the gear meshing system.

3. When the gear system was operated in the range of 33 to 100 percent of full load and at 83 and 100 percent of full speed, the losses, and therefore the efficiency, were significantly effected by the windage to the point where the losses due to windage was nearly equal to or exceeded that of the gear meshing (rolling and sliding) losses.

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